US006200111B1

(12) **United States Patent**
Foss

(10) **Patent No.: US 6,200,111 B1**
(45) **Date of Patent: Mar. 13, 2001**

(54) **CONTROLLABLE HIGH VOLUME POSITIVE DISPLACEMENT PUMP**

(76) Inventor: **John F. Foss**, 4731 Nakoma Dr., Okemos, MI (US) 48864

(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 0 days.

(21) Appl. No.: **09/239,120**

(22) Filed: **Jan. 28, 1999**

(51) **Int. Cl.**[7] **F04B 7/00**

(52) **U.S. Cl.** **417/515**; 137/50; 137/628; 251/215

(58) **Field of Search** .... 417/515, 286, 417/297; 137/627.5, 50, 628; 251/215; 74/25, 30, DIG. 7, 423; 474/69; 123/90.44

(56) **References Cited**

U.S. PATENT DOCUMENTS

1,060,803	*	5/1913	Welch	74/3
2,889,783	*	6/1959	Woydt	103/169
3,266,435	*	8/1966	Smith	103/170
4,787,822	*	11/1988	Bennett	417/2
5,375,625	*	12/1994	Reynolds	137/625.69
5,690,567	*	11/1997	DeNijs	474/73

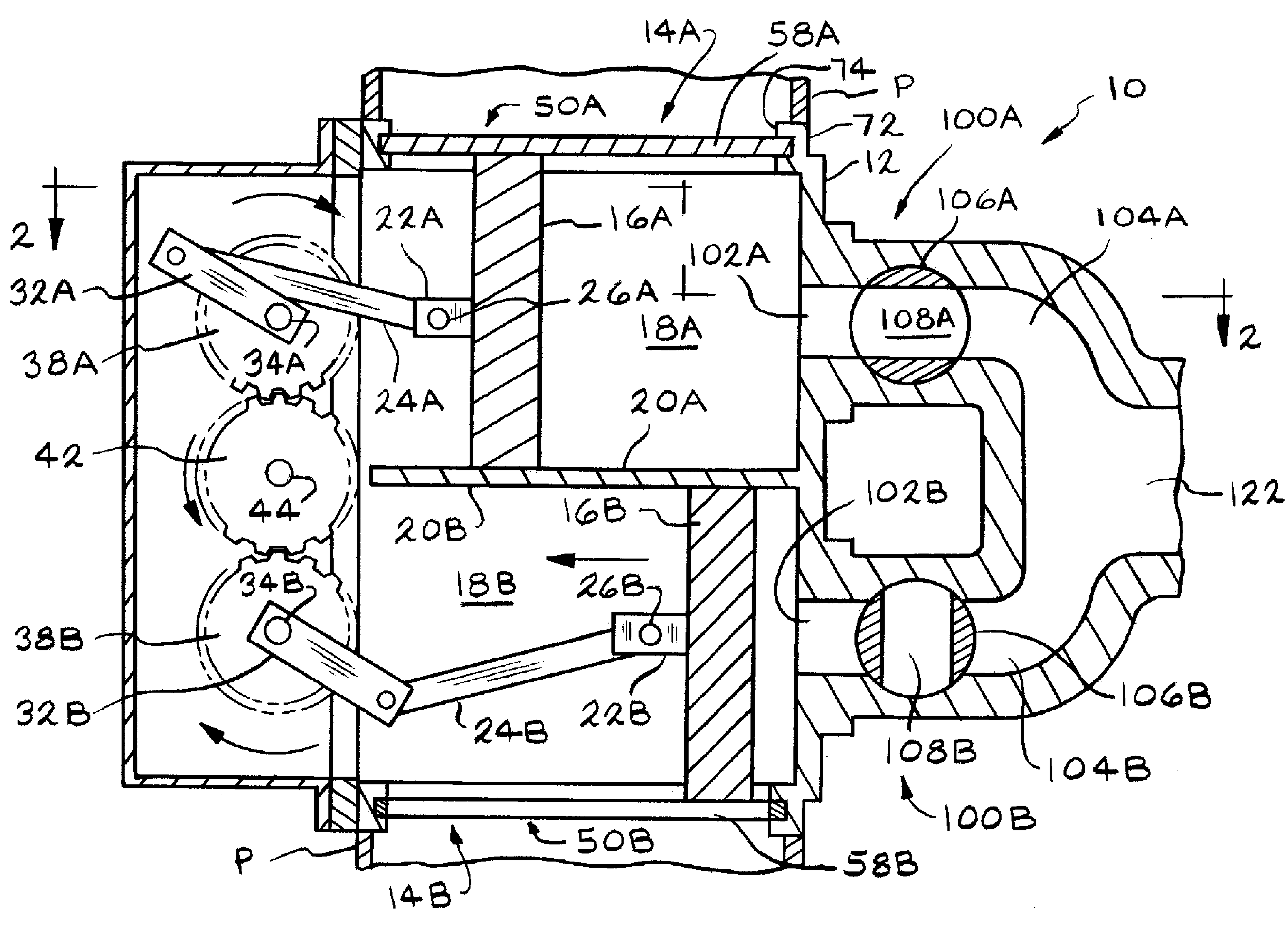
* cited by examiner

Primary Examiner—Timothy S. Thorpe
Assistant Examiner—Ehud Gartenberg
(74) *Attorney, Agent, or Firm*—Brinks Hofer Gilson & Lione

(57) **ABSTRACT**

A controllable positive displacement pump for liquids and gases which may contain dispersed phase particles includes a pair of pistons operating out of phase. Each of the pistons is associated with an inlet valve which includes a fixed grating defining slots which extend parallel to piston travel and a complementarily configured grating which reciprocates transversely in timed relationship with the piston to control the influx of fluid. A rotary outlet valve is also associated with each piston and includes a rotating member disposed transversely across the cylinder head and which rotates in synchronism to open a through, radial port in the member in timed relationship to the piston travel. The positive displacement pump finds particular application to transport powder paint.

20 Claims, 9 Drawing Sheets



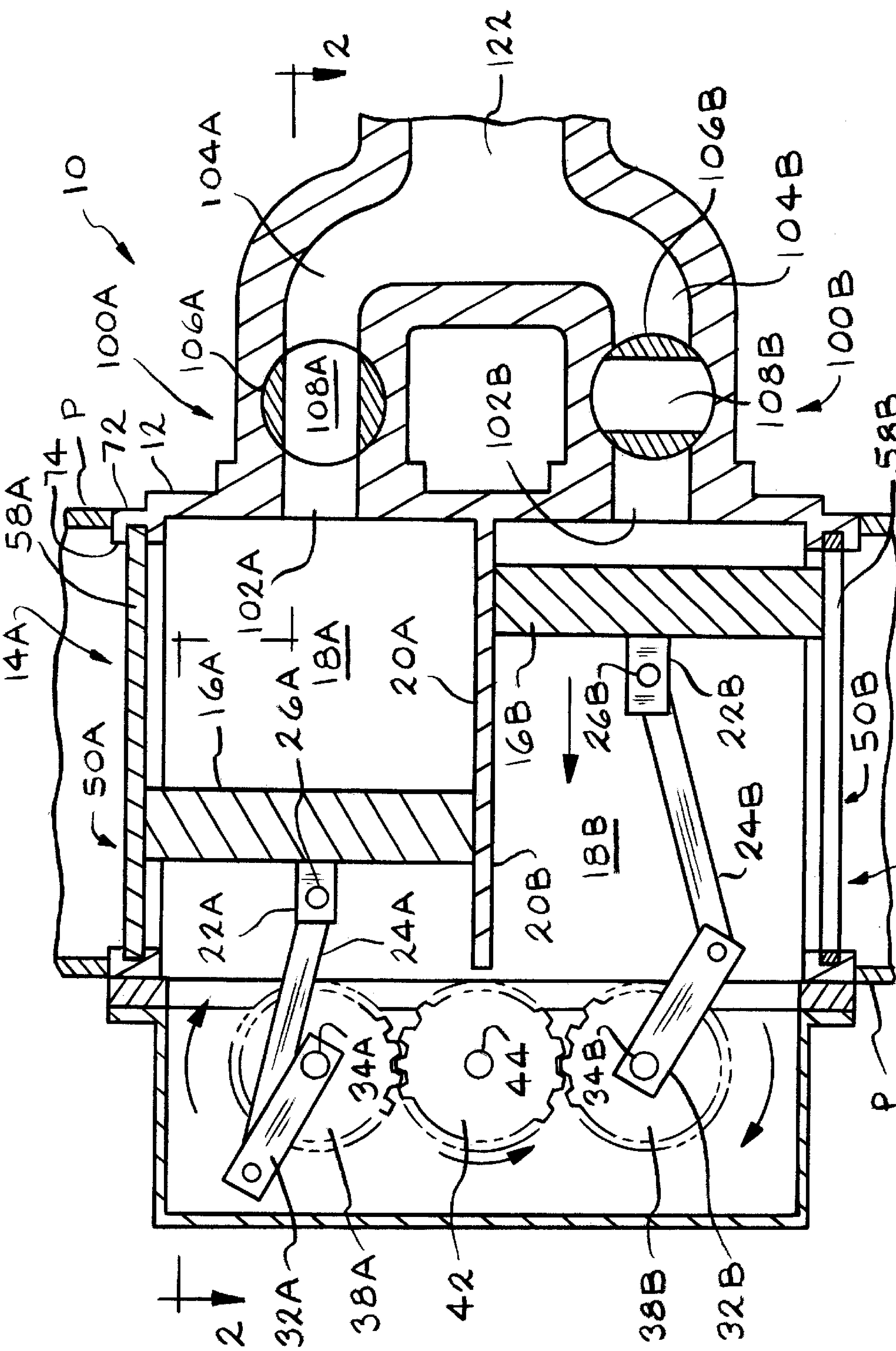

10
2
122
104A
106B
104B
100A
106A
108A
102B
108B
100B
P
74
72
12
58A
58B
14A
102A
18A
16A
26A
20A
16B
26B
22B
50A
50B
24B
18B
20B
22A
24A
14B
34A
44
34B
42
32A
38A
38B
32B
2
P

FIG. 1

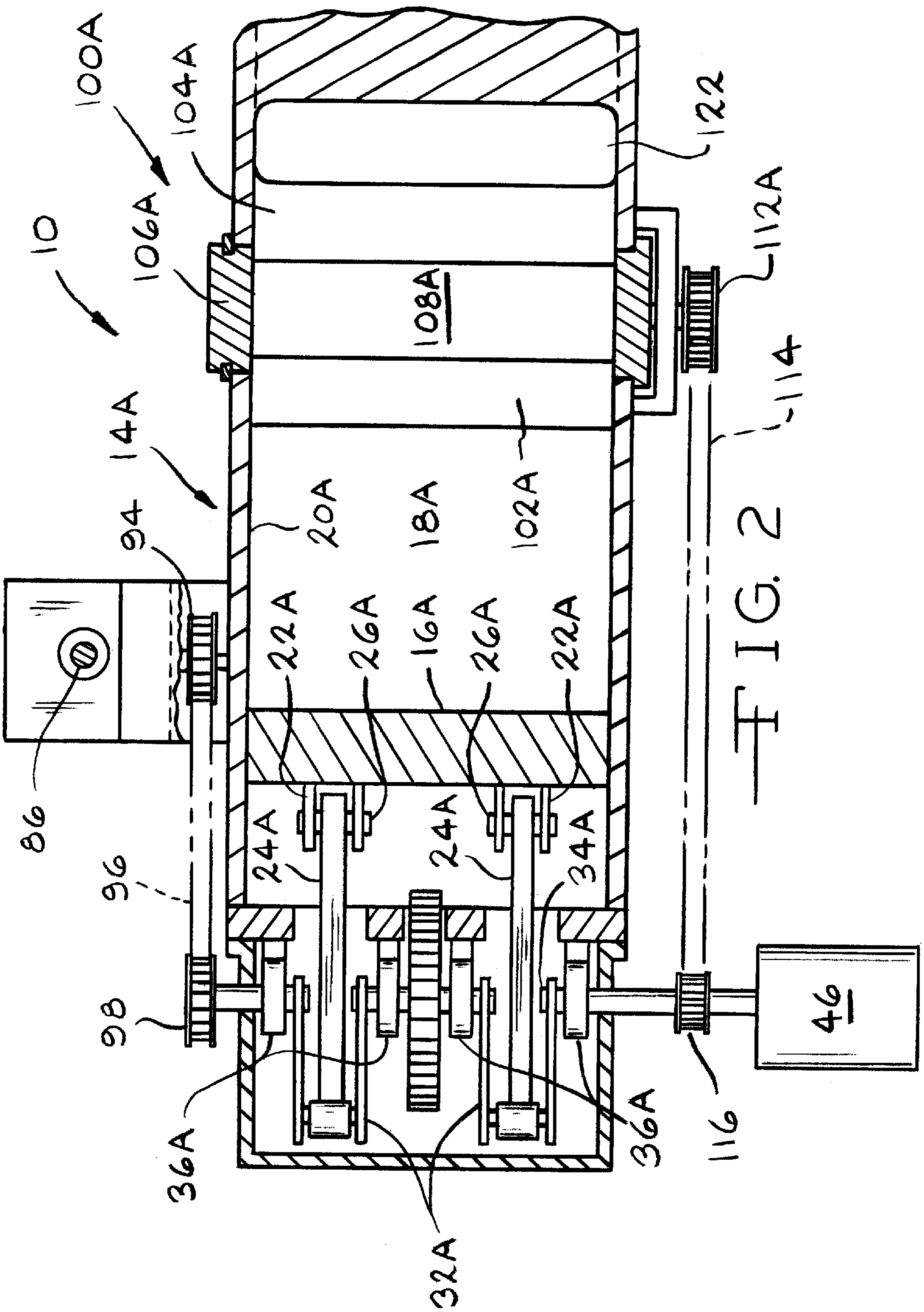

100A
104A
122
10
106A
112A
108A
114
14A
20A
18A
102A
94
22A
26A
16A
26A
22A
86
24A
24A
34A
96
98
46
36A
116
36A
36A
32A

FIG. 2

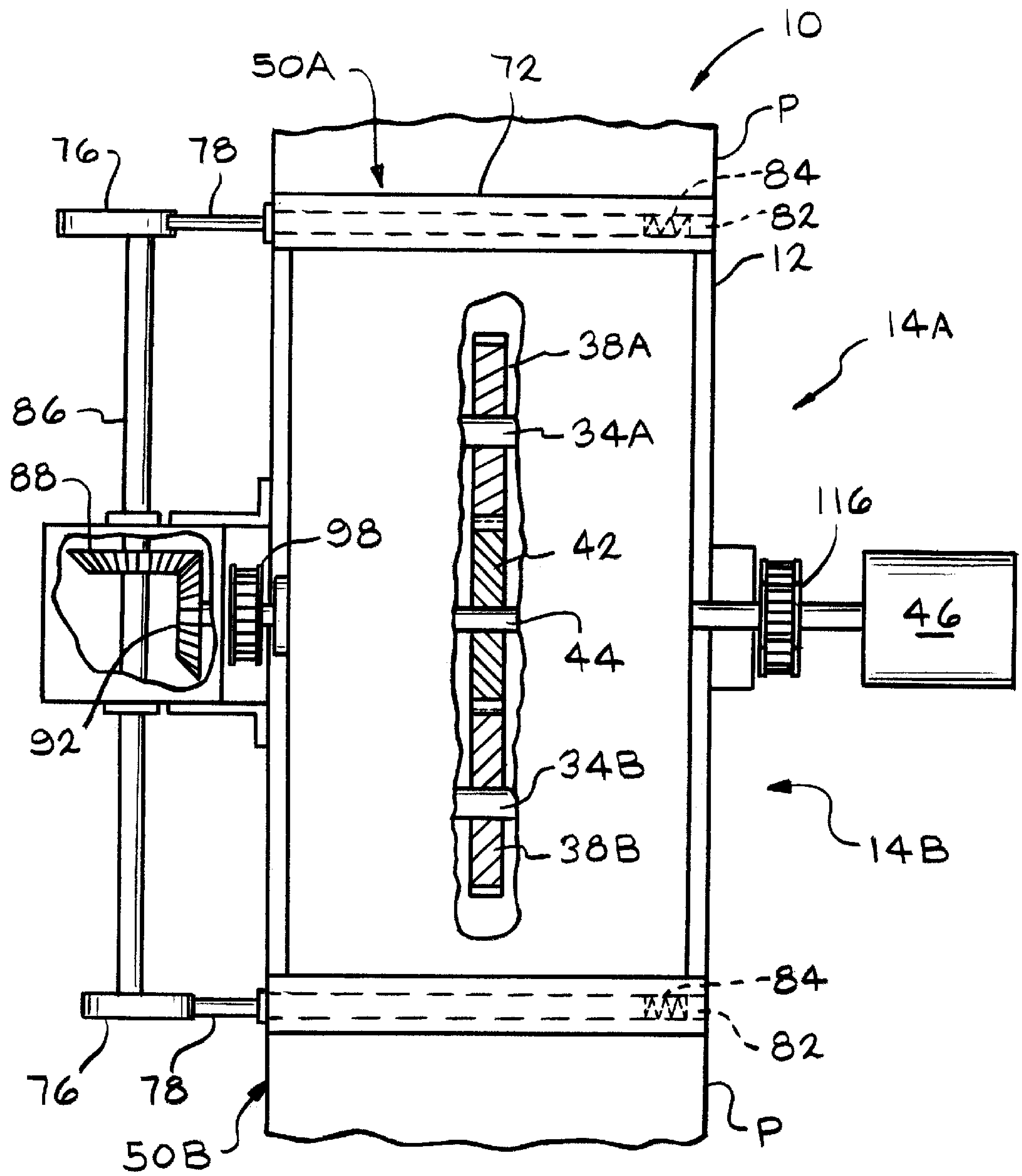

10
50A
72
P
76
78
84
82
12
38A
14A
86
34A
88
116
98
42
46
44
92
34B
14B
38B
84
82
76
78
P
50B

FIG. 3

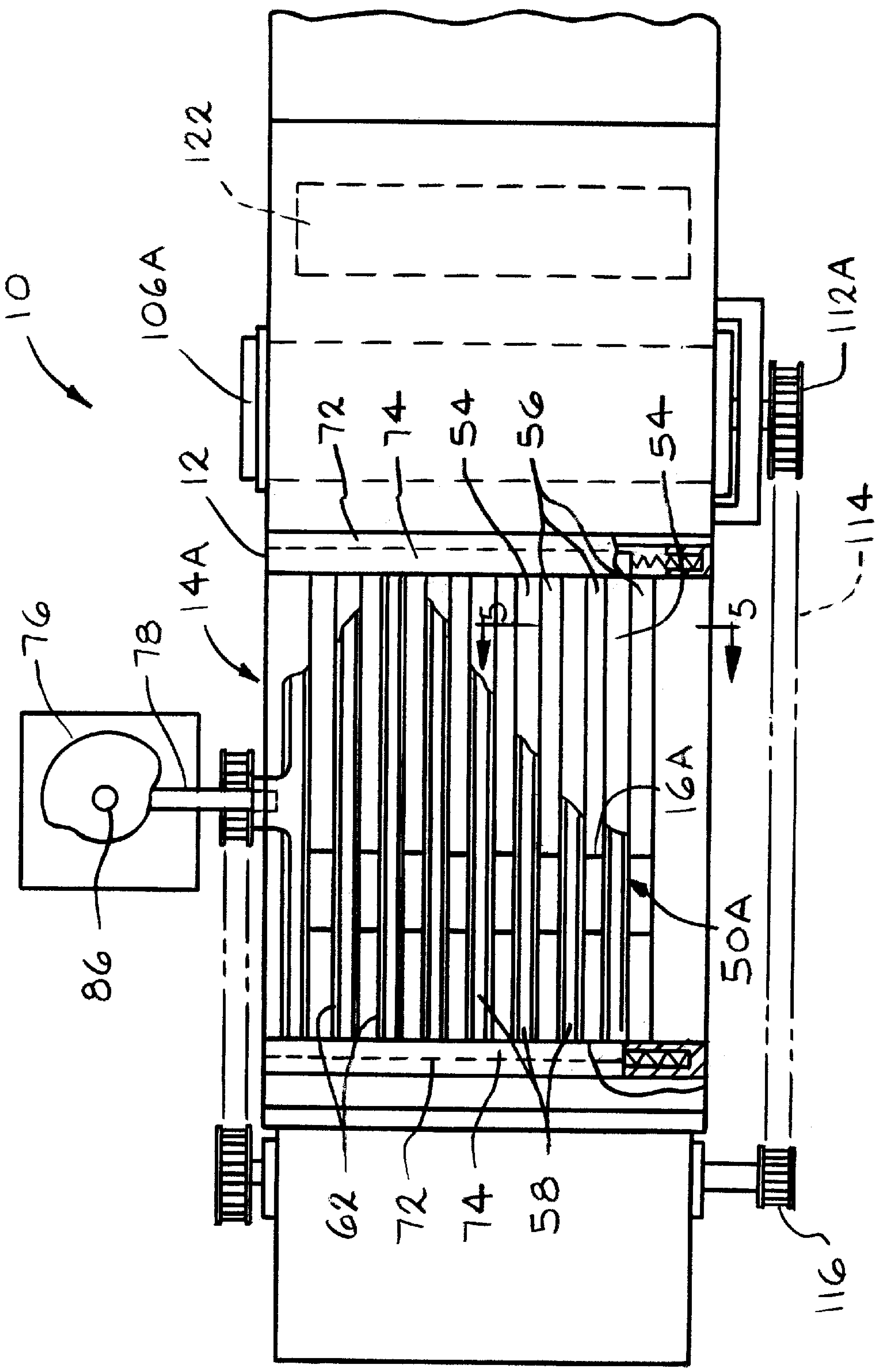

10
12
14A
16A
50A
54
54
56
58
62
72
72
74
74
76
78
86
106A
112A
114
116
122
5
5

FIG. 4

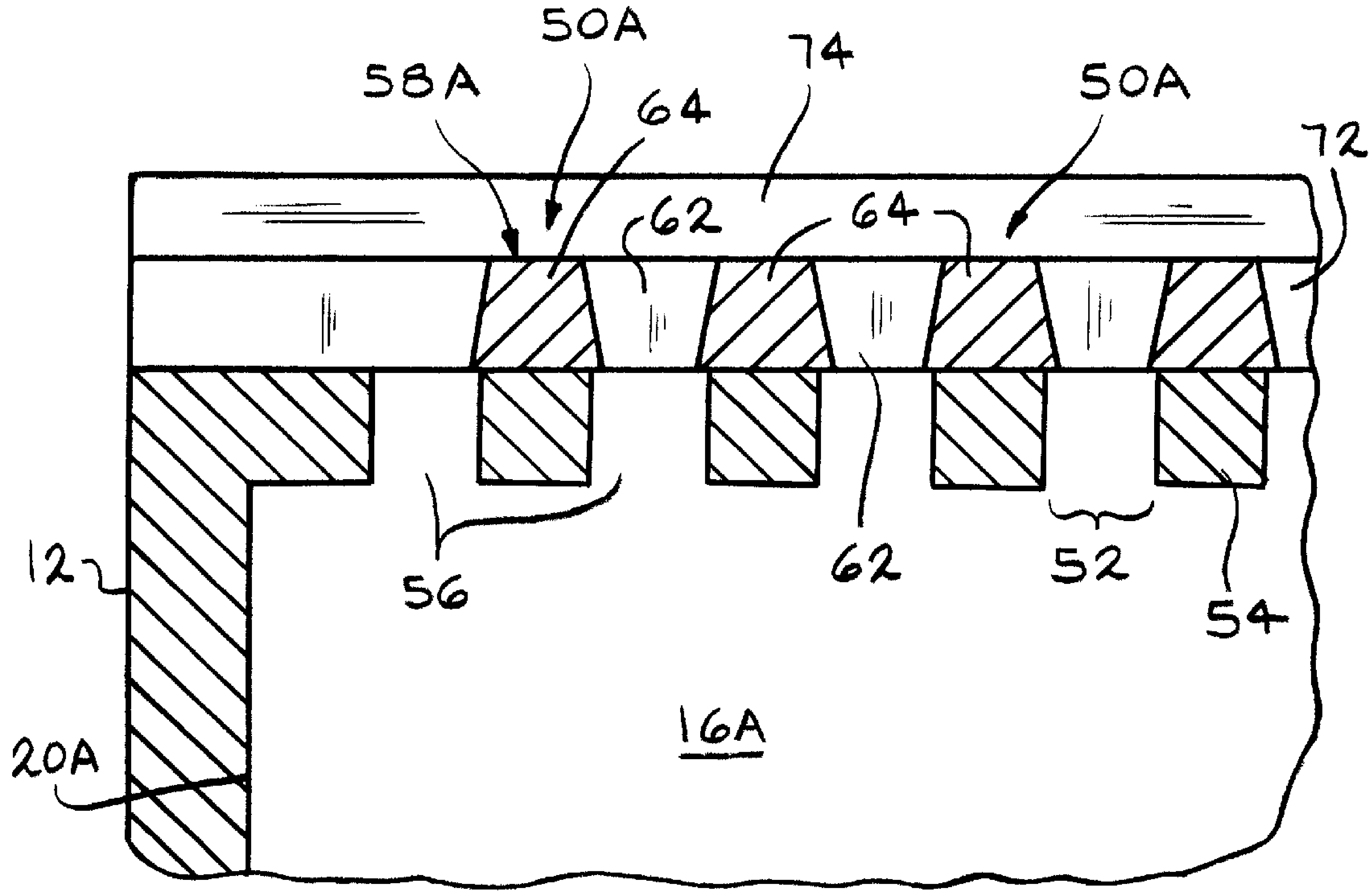


FIG. 5

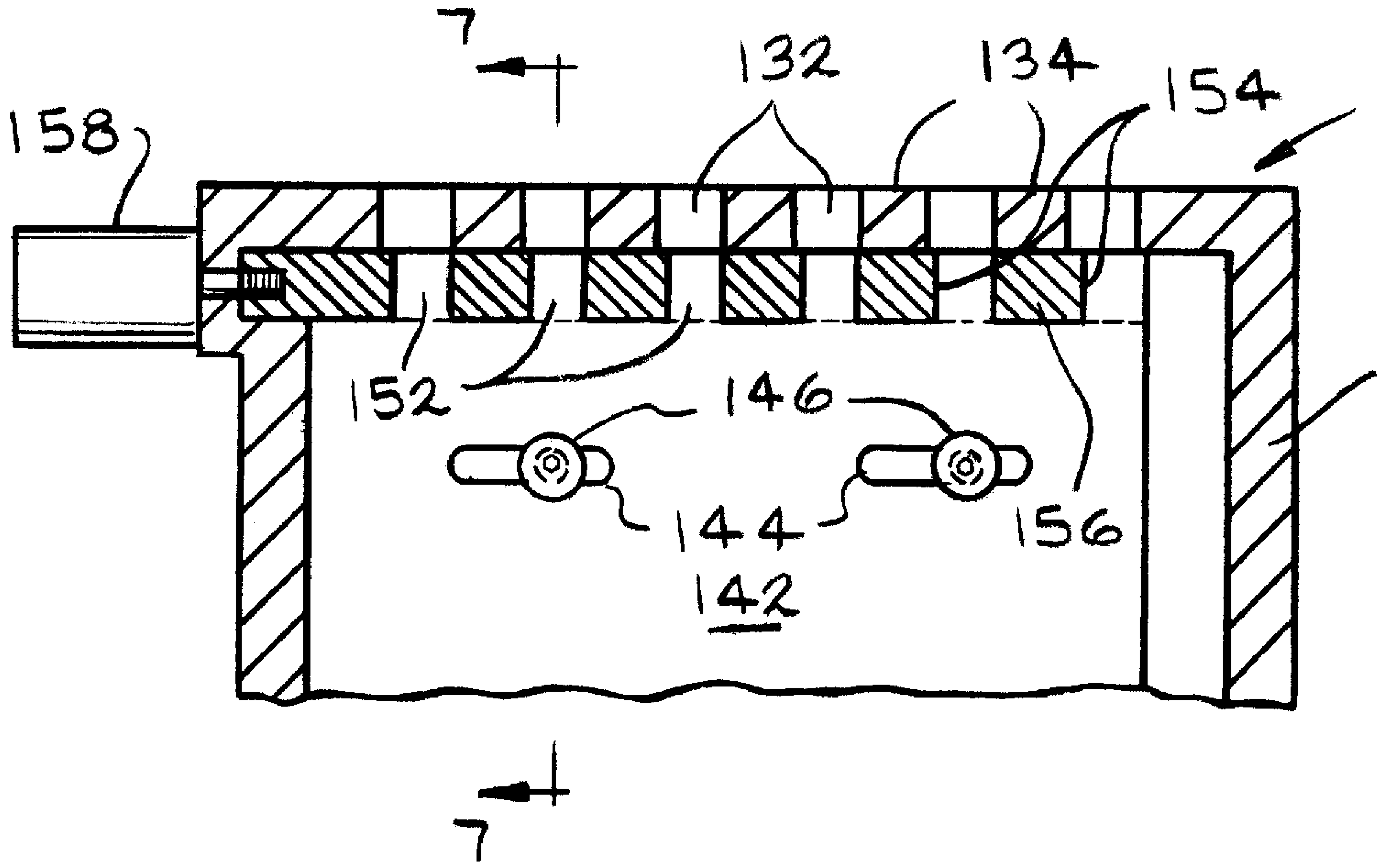

7
132
134
154
130'
158
12'
152
146
144
156
142
7

FIG. 6

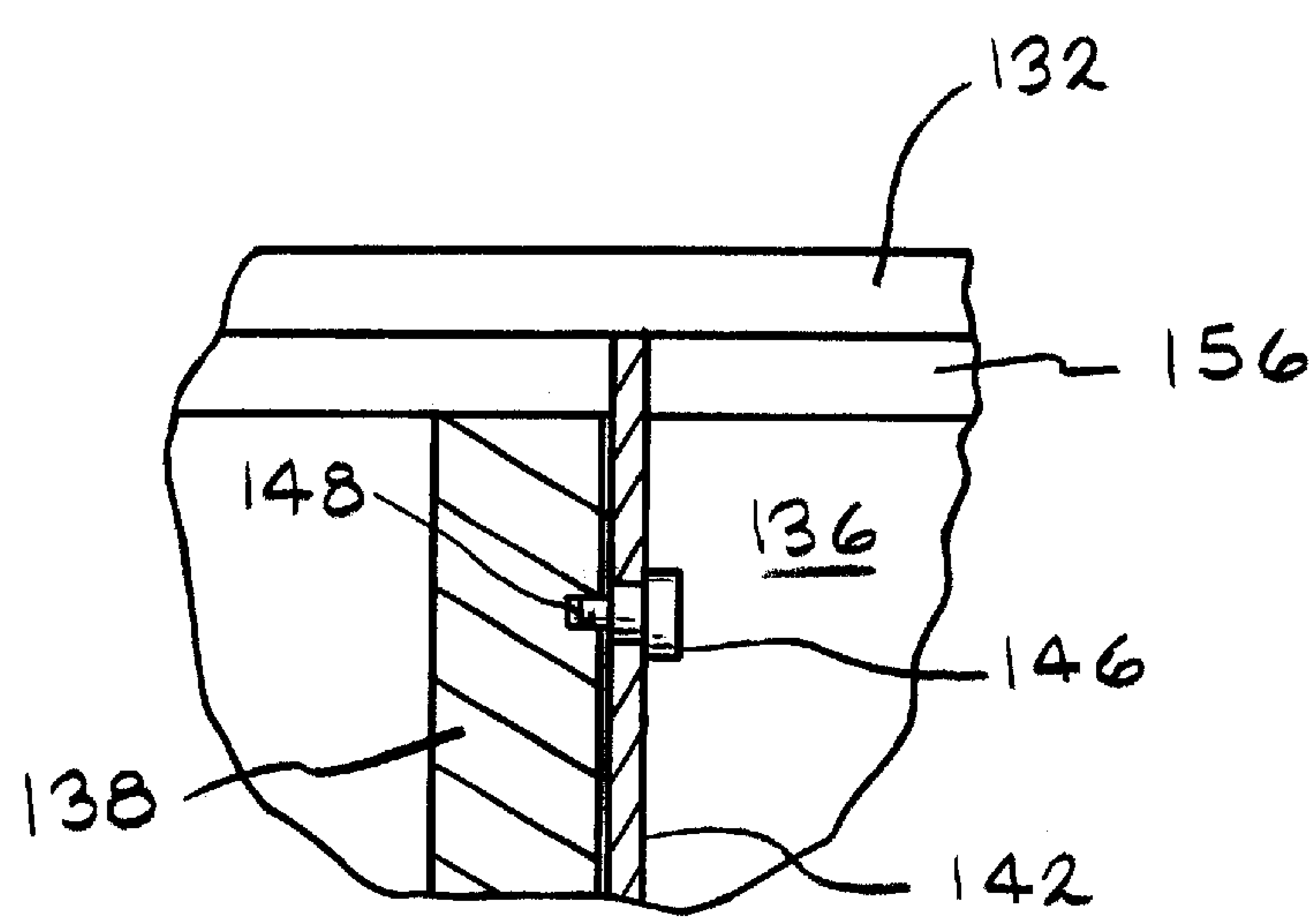

132
156
148
136
146
138
142

FIG. 7

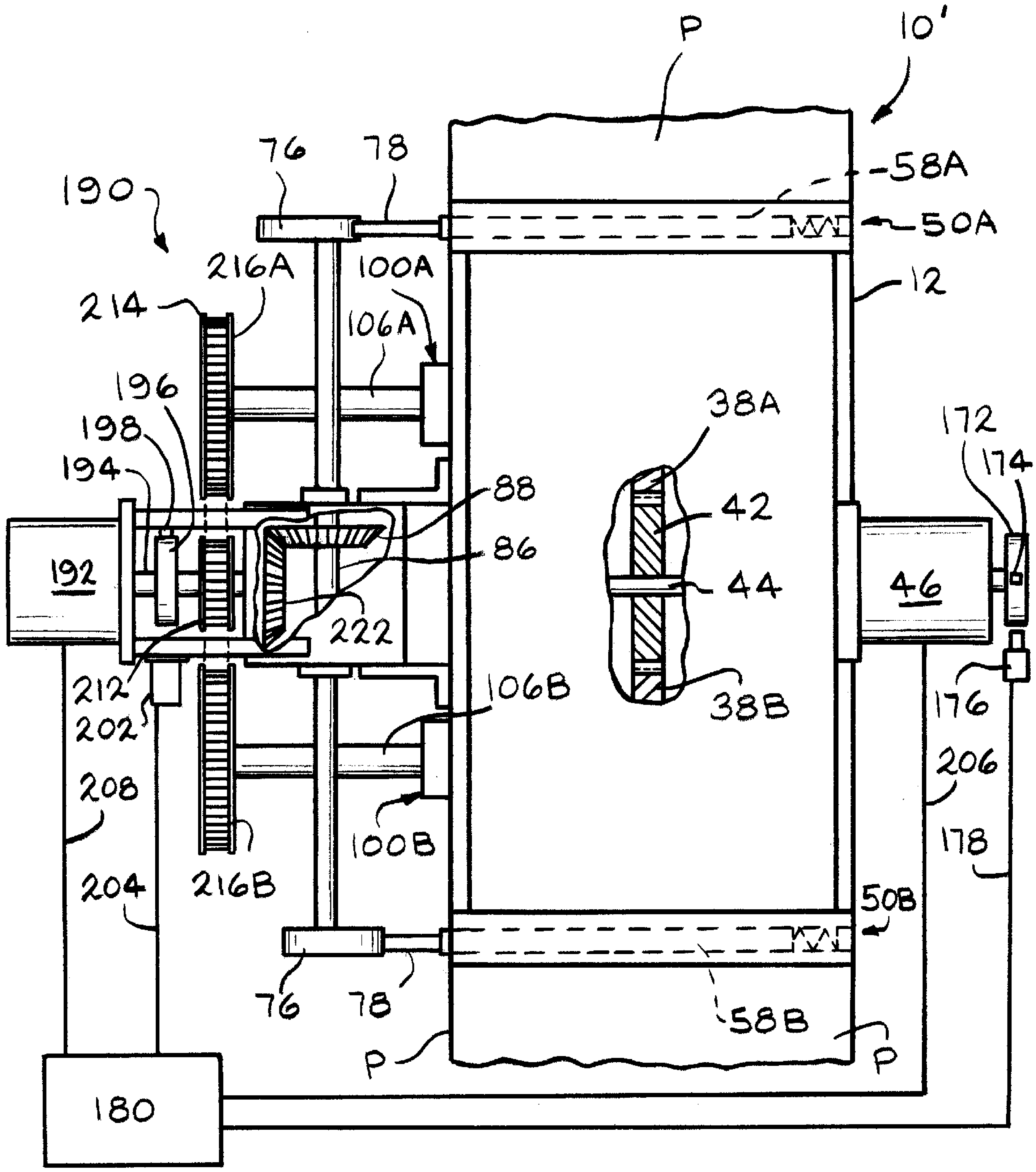

P
10'
76
78
58A
50A
190
216A
100A
12
214
106A
196
38A
198
172
174
194
88
42
192
86
44
46
222
212
202
106B
38B
176
208
206
100B
178
204
216B
50B
76
78
58B
P
P
180

FIG. 8

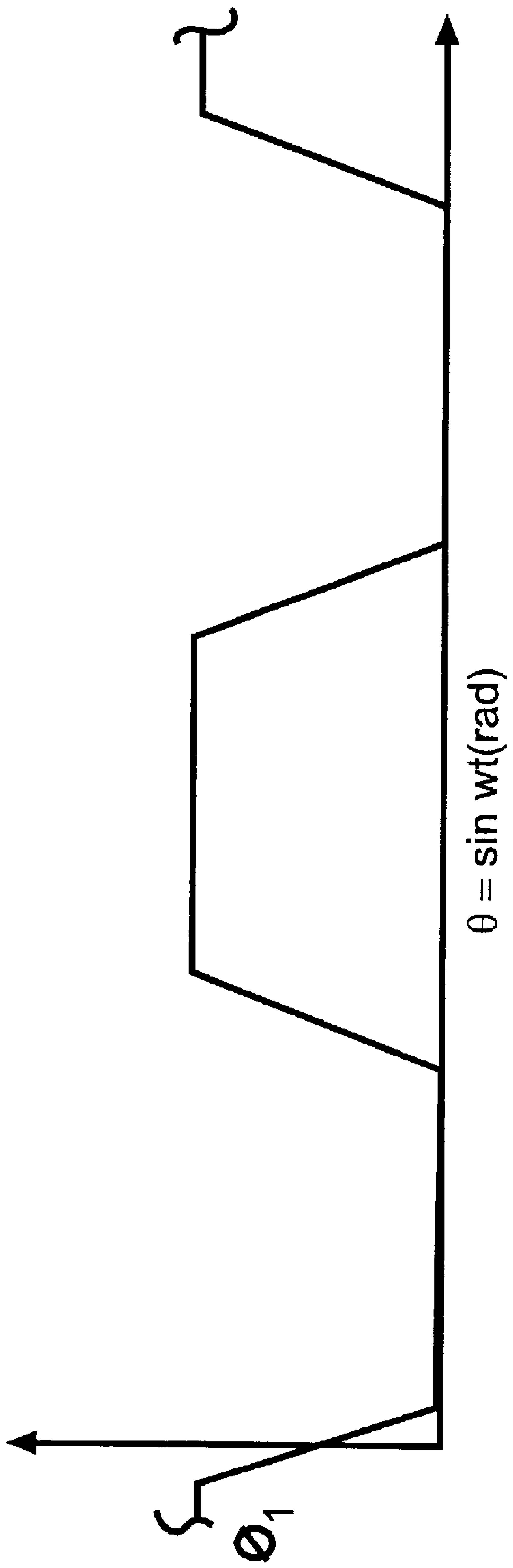


FIG. 9

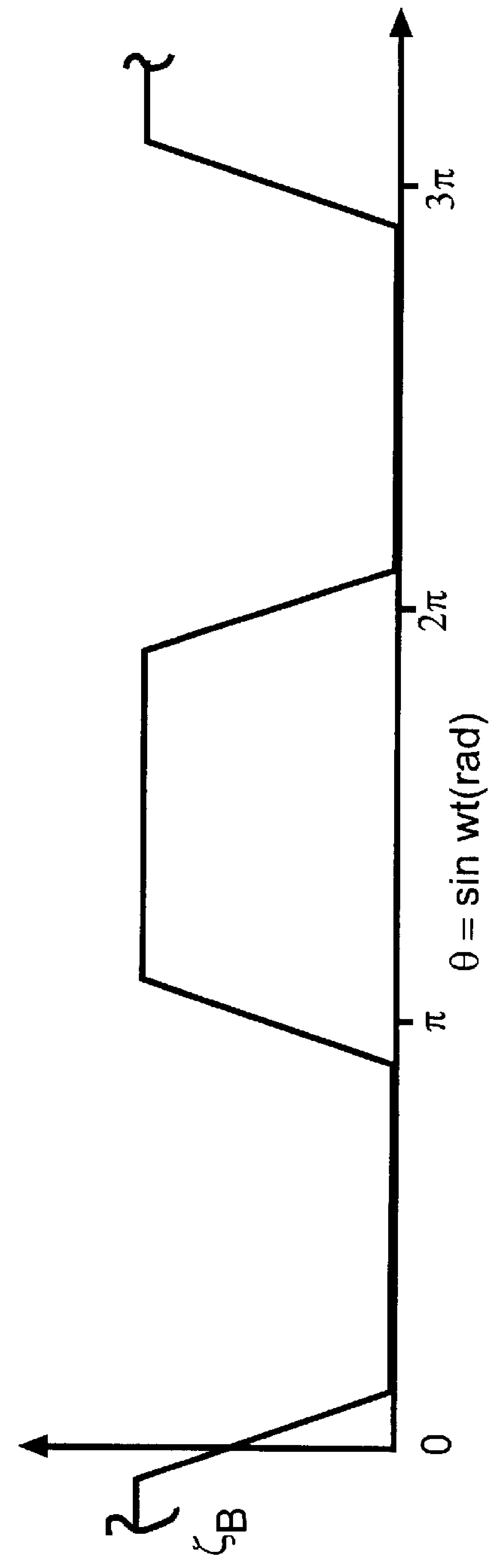


FIG. 10

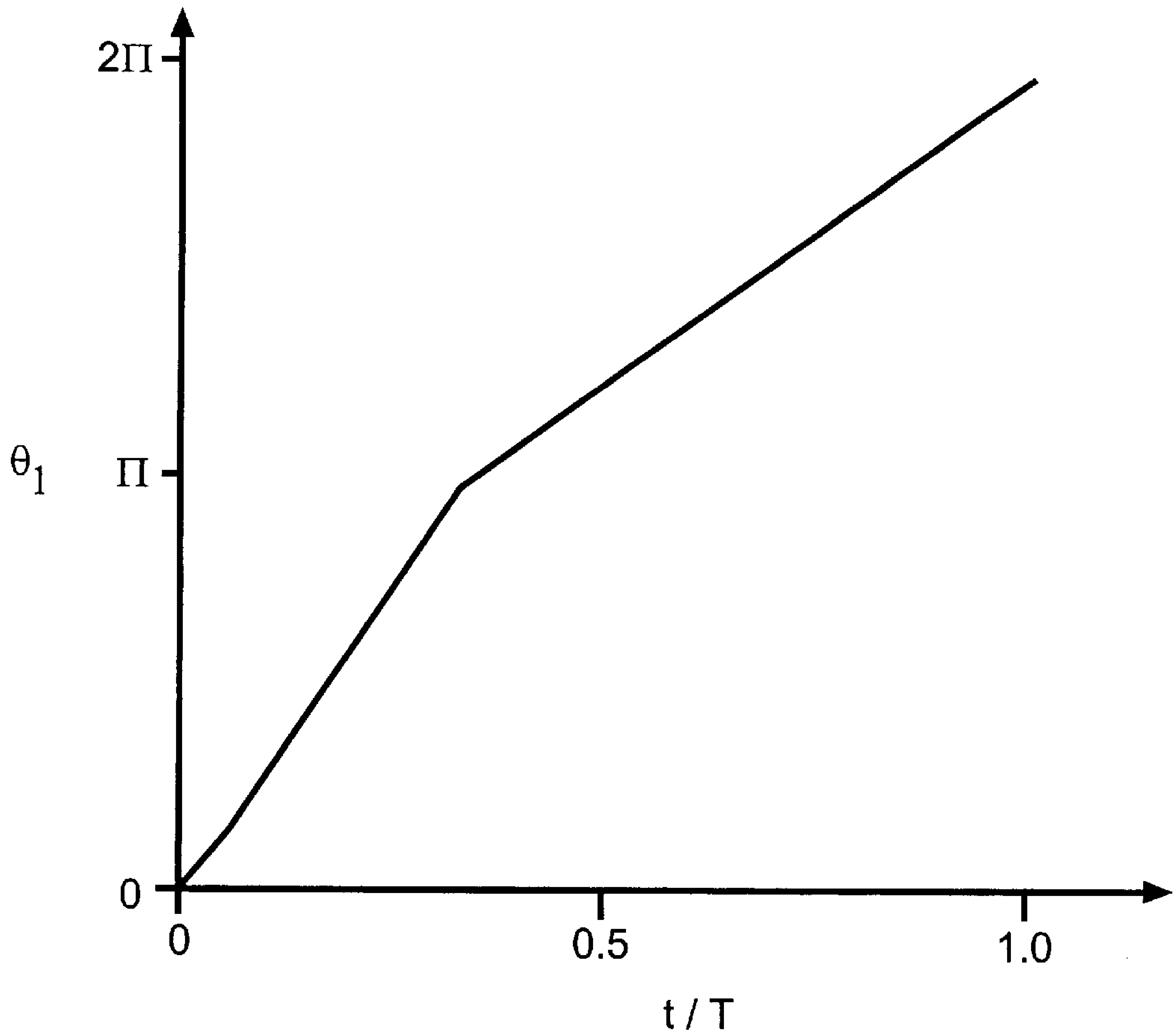


FIG. 11

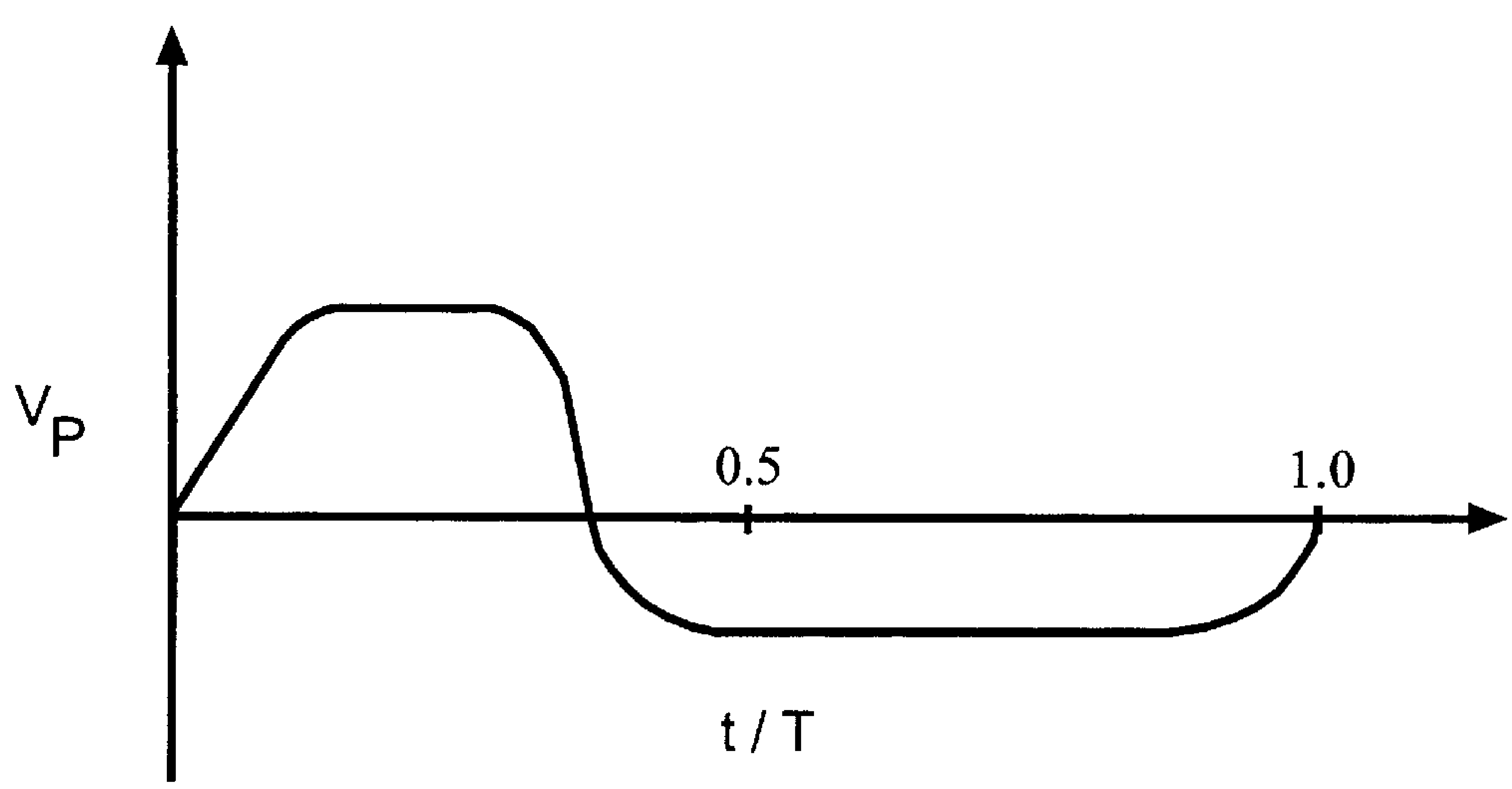


FIG. 12

CONTROLLABLE HIGH VOLUME POSITIVE DISPLACEMENT PUMP

BACKGROUND OF THE INVENTION

The invention relates generally to positive displacement pumps for liquids and gases and more specifically to a controllable positive displacement pump having a pair of pistons operating out of phase and specially configured and controllable inlet valves.

Positive displacement pumps for liquids and gases typically include one or more piston and cylinder assemblies and associated inlet and outlet valves which control the flow of pumped fluid into and out of the cylinders. Such pumps are typically capable of relatively high pressure rise operation. A drawback of such positive displacement pumps is that both the inflow and outflow are distinctly pulsatile in character and, especially with high pressure pumps, the flow rates are generally relatively small.

Furthermore, the ability to adjust pressure and flow rates with such pump can be problematic. Typically, of course, flow rates may be adjusted simply by reducing the speed of the pump. However, such a speed reduction to reduce output flow rate is typically accompanied by a reduction in the output pressure as well.

It is apparent from the foregoing that a positive displacement pump which addresses the problems of output pulsation and controllable flow characteristics would represent an improvement over currently available devices.

SUMMARY OF THE INVENTION

The present invention is directed to a controllable, high volume positive displacement pump which provides a flow rate having temporal fluctuations which are a smaller fraction of the time mean value than those of conventional positive displacement pumps, is characterized as a higher flow rate, smaller pressure rise device in comparison to conventional positive displacement pumps and readily permits independent control of the intake and exhaust valve phasing and cycle times.

A controllable positive displacement pump for liquids and gases which may contain dispersed phase particles includes a pair of pistons operating out of phase which provide pumped fluid to a common output. Each of the pistons is associated with an inlet valve which includes a fixed grating defining slots which extend parallel to piston travel and a complementarily configured grating which reciprocates transversely in timed relationship with the piston to control the influx of fluid. A rotary outlet valve is also associated with each piston and includes a rotating member disposed transversely across the cylinder head which rotates in synchronism to open a through, radial port in the member in timed relationship to the piston travel. The phase relationship between the operation of the inlet and outlet valves and the respective pistons may be adjusted by using either independent drive mechanisms to these components or incorporating mechanical phase adjusting devices in the unitary drive mechanism. The positive displacement pump finds particular application to transport powder paint and in heating, ventilating and air conditioning (HVAC) apparatus.

Thus it is an object of the present invention to provide a controllable, positive displacement pump.

It is a further object of the present invention to provide a controllable, positive displacement pump suitable for applications such as transport of powder paint and in HVAC apparatus.

It is a still further object of the present invention to provide a controllable, positive displacement pump wherein inlet and outlet valves operate in synchronism with reciprocating pistons.

It is a still further object of the present invention to provide a controllable, positive displacement pump wherein the phase relationships of the inlet and outlet valves may be adjusted relative to the reciprocating pistons.

Further objects and advantages of the present invention will become apparent by reference to the following description of the preferred and alternate embodiments and appended drawings wherein like reference numbers refer to the same component, element or feature.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** is a side, elevational view in full section of a controllable positive displacement pump according to the present invention;

FIG. **2** is a full, sectional view of one of the piston and cylinder assemblies of a controllable positive displacement pump according to the present invention take along line **2—2** of FIG. **1**;

FIG. **3** is an end, elevational view with portions broken away of a controllable positive displacement pump according to the present invention;

FIG. **4** is a top, plan view of a controllable positive displacement pump according to the present invention;

FIG. **5** is a fragmentary, sectional view of an inlet valve of a controllable positive displacement pump according to the present invention taken along line **5—5** of FIG. **4**;

FIG. **6** is a fragmentary, sectional view of a first alternate embodiment piston and inlet valve configuration according to the present invention;

FIG. **7** is a fragmentary, sectional view of the first alternate embodiment piston and inlet valve configuration according to the present invention taken along **7—7** of FIG. **6**;

FIG. **8** is an end, elevational view with portions broken away of a second alternate embodiment controllable positive displacement pump according to the present invention having independent phase adjustable drives for the pistons and valves;

FIG. **9** is a timing diagram for the upper piston and cylinder assembly of a controllable positive displacement pump according to the present invention at maximum flow rate;

FIG. **10** is a graph presenting the position of the upper inlet valve sliding plate of a controllable positive displacement pump according to the present invention as a function of the crank angle for a maximum flow rate condition;

FIG. **11** is a graph presenting the crank angle position versus time over one operating cycle of a controllable positive displacement pump according to the present invention; and

FIG. **12** is a graph presenting an asymmetric driving condition of a controllable positive displacement pump according to the present invention.

DESCRIPTION OF THE PREFERRED AND ALTERNATE EMBODIMENTS

Referring now FIGS. **1** and **2**, a controllable, high volume, positive displacement pump according to the present invention is illustrated and generally designated by the reference number **10**. The positive displacement pump **10** includes a housing **12** which is preferably cast metal and includes various apertures, surfaces and ports which cooperate with other features of the invention. Specifically, the positive displacement pump **10** includes an upper or first piston and cylinder assembly **14**A and a lower or second piston and cylinder assembly **14**B. The upper piston and cylinder assembly **14**A and the lower piston and cylinder assembly **14**B are substantially identical and the upper piston and cylinder assembly **14**A includes a first preferably rectangular piston **16**A disposed within a complementary first rectangular cylinder **18**A defined by a first rectangular cylinder wall **20**A. The piston **16**A includes a first pair of devises **22**A which receive a respective first pair of connecting rods **24**A which are pinned to the devises by a respective first pair of retaining pins **26**A. The first pair of connecting rods **24**A are in turn pivotally received on a first pair of respective cranks **32**A of an first crankshaft **34**A. The first crankshaft **34**A is supported for a rotation in a plurality of first bearings **36**A which may define either standard journal bearings or anti-friction devices such as ball bearing assemblies (not illustrated). Secured generally centrally to the crankshaft **34**A is a first driven pinion gear **38**A which is in constant mesh with a drive pinion gear **42**. The drive pinion gear **42** is secured to a transverse drive shaft **44** and driven by a prime mover such as a variable speed electric motor **46**.

The lower or second piston and cylinder assembly **14**B is in all mechanical respects the same as the upper piston and cylinder assembly **14**A except that it operates 180° out of phase with the first or upper piston and cylinder assembly **14**A. Thus, it includes a second piston **16**B, a second cylinder **18**B, a second cylinder wall **20**B, second pairs of devises **22**B, a second pair of connecting rods **24**B, a second pair of retaining pins **26**B, a second pair of cranks **32**B, a second crankshaft **34**B, second bearings and a second driven pinion gear **38**B. It will be appreciated that the first cranks **32**A and the second cranks **32**B are arranged 180° out of phase from one another as illustrated in FIG. **1**.

Turning now to FIGS. **1**, **3**, **4** and **5**, each of the first and second piston and cylinder assemblies **14**A and **14**B includes a respective inlet valve assembly **50**A and **50**B. The upper or first inlet valve assembly **50**A is disposed adjacent the upper cylinder wall **20**A of the first or upper piston and cylinder assembly **14**A and the second or lower inlet valve assembly **50**B is disposed adjacent the lower cylinder wall **20**B of the second or lower piston and cylinder assembly **14**B. The inlet valve assemblies **50**A and **50**B, but for their locations, are mechanically identical with the exception that once again, they operate 180° out of phase from one another. Hence, only the first or upper inlet valve assembly **50**A will be described, it being understood that the same description applies to the second or lower inlet valve assembly **50**B.

Formed in the upper portion of the cylinder wall **20**A are a plurality of longitudinally extending slots **52** defined by longitudinally extending bars **54**. Received within the slots **52** defined by the bars **54** are complementarily configured rectangular teeth or projections **56** which form and define the upper edge of the piston **16**A. Transversely, slidingly disposed immediately above and adjacent the grating defined by the bars **54** is a complementarily configured valve plate **58** having a plurality of longitudinally extending slots **62** having generally trapezoidal cross sections which are defined by a complementarily configured arrangement of trapezoidally shaped bars **64**. The adjacent, facing surfaces of the bars **54** and **64** as well as the edges of the teeth **56** of the piston **16**A are preferably uniformly polished or finished such that they slide smoothly against one another in intimate contact to provide a suitable fluid tight seal. The valve plate **58** is slidingly retained upon the top of the piston and cylinder assembly **14**A by a pair of parallel, right angle (L-shaped) guides **72** having overhanging lips **74**.

The inlet valve assembly **50**A and specifically the valve plate **58** is reciprocated in proper timed relationship with the motion of the first piston **16**A by a cam **76** and a follower **78** and is open when the first piston **16**A is on its intake stroke and is closed when the first piston **16**A is on its compression stroke.

Each of the guides **72** includes a closed end portion **82** which receives a compression spring **84** which biases the valve plate **58**A and provides restoring force to return it to the left is illustrated in FIG. **3** to achieve proper reciprocation of the plate **58** as will readily be appreciated. The cam **76** is coupled to a drive shaft **86** to which a bevel gear **88** is secured. The bevel gear **88**, in turn, is driven by a matching bevel gear **92**. As shown in FIGS. **2** and **3**, the bevel gear **92** is coupled to a driven pinion **94** and, through a timing belt or chain **96**, to a drive pinion **98** secured to the drive shaft **44**. The drive ratio from the drive shaft **44**, through the timing belt **96** and through the bevel gears **88** and **92** is 1:1 and thus the cams **76** and the valve plates **58**A and **58**B operate in synchronism with the motion of the pistons **16**A and **16**B.

Returning now to FIGS. **1** and **2**, each of the piston and cylinder assemblies **14**A and **14**B also includes respective rotary outlet valve assemblies **100**A and **100**B. The upper or first rotary valve assembly **100**A and lower or second rotary valve assembly **100**B are mechanically identical and again, the only difference being operational in that they typically operate 180° out of phase from one another, in synchronism with the motion of the associated respective pistons **16**A and **16**B. Accordingly, only the first or upper outlet valve assembly **100**A will be described. In the headwall of the cylinder wall **20**A is a slot **102**A defined by the housing **12** which preferably extends substantially the full width of the piston **16**A and the cylinder **18**A. The slot **102**A opens into a passageway **104**A communicating with a rotary valve body **106**A. Disposed across the full width of the slot **102**A and between the slot **102**A and the passageway **104**A is a rotary valve body **106**A having a through radial passageway **108**A defining a height substantially equal to the height of the slot **102**A and the passageway **104**A. The remaining material in the valve body **106**A adjacent the passageway **108**A when rotated 90° is sufficiently wide to fully close off the passageway **104**A from the cylinder **18**A and the slot **102**A. The rotary valve body **106**A is coupled to a drive pinion **112**A which is driven through a timing belt **114** from a pinion **116** secured to the crankshaft **34**B. It will be noted that the driven pinion **112**A has a diameter twice as large as the drive pinion **116** and this configuration effects a 2:1 speed reduction such that the rotary valve body **106**A rotates one revolution for every two revolutions of the crankshaft **34**A. Flow through a slot **102**B and through a passageway **108**B in the rotary valve body **106**B is similarly controlled. As illustrated in FIG. **1**, the passageway **104**A merges with a similar passageway **104**B from the second or lower piston and cylinder assembly **14**B and thus fluid flow from the cylinders **18**A and **18**B merge in a common outlet passageway **122**.

Turning now to FIGS. **6** and **7**, a first alternate embodiment inlet valve assembly **130** which relates to the interface and seal between the pistons **16**A and **16**B and their respective inlet valve assemblies **50**A and **50**B is illustrated. The first alternate embodiment inlet valve assembly **130** includes an alternate embodiment housing **12'** having a plurality of longitudinal slots **132** defined by a plurality of parallel, longitudinal bars **134**. The bars **134** are preferably integrally formed with the housing **12'** when it is fabricated, for example, by casting. The slots **132** provide fluid communication between the ambient and a cylinder **136** defined by the housing **12'**. Disposed within the cylinder **136** is a reciprocating piston **138**. It will be appreciated that but for the differences about to be described, the piston **138** is the same as the first and second pistons **16**A and **16**B, respectively, described above with regard to the preferred embodiment positive displacement pump **10**. The piston **138** includes a transversely slidable plate **142** having at least a pair of elongate slots **144** through which a respective pair of shoulder bolts **146** are disposed. The shoulder bolts **146** mount into suitably configured blind apertures **148** in the piston **138** and retain and slidably position the plate **142** upon the face of the piston **138**. The plate **142** includes a plurality of rectangular teeth or projections **152** which are received within complementarily configured elongate slots **154** in a valve plate **156**.

The valve plate **156** is transversely reciprocated through the agency of a cam and follower assembly **158** which is similar in all respects to the cam **76** and follower **78** of the preferred embodiment pump **10**. As the piston **138** reciprocates within the cylinder **136**, the alternate embodiment inlet valve assembly **130** alternately allows ingress of pumped fluid and compression thereof by opening and closing in synchronism with the piston **138** by first placing the slots **132** and **154** in alignment as illustrated in FIG. **6** and then closing off the slots **132** with the bars of the valve plate **156** as it reciprocates. The plate **142** thus provides an appropriate seal while accommodating the interfacial motion of the reciprocating piston **138** along its axis and the transversely oriented reciprocating motion of the valve plate **156**.

Referring now to FIG. **8**, a second alternate embodiment of a controllable positive displacement pump **10'** is illustrated. The positive displacement pump **10'** is identical to the preferred embodiment positive displacement pump **10** in all respects with regard to the mechanical configuration of the piston and cylinder assemblies **14**A and **14**B, the inlet valve assemblies **50**A and **50**B and the rotary outlet valve assemblies **100**A and **100**B. The second alternate embodiment pump assembly **10'** differs from the preferred embodiment pump **10** in the drive mechanisms for the components and more particularly to the capability of the drive mechanisms to adjust the phasing therebetween. Accordingly, the second alternate embodiment pump **10'** includes the drive motor **46** which directly drives the shaft **44**, the drive pinion gear **42** and the driven pinion gears **38**A and **38**B. Attached to the drive shaft **42** is a tone wheel **172** having one or more teeth **174** which are configured to permit angular or phase resolution of the position of the shaft **42** and thus of the pistons **16**A and **16**B. A sensor such as a Hall effect or variable reluctance sensor **176** is disposed in sensing relationship with the tone wheel **172** and provides a signal through a line **178** to a controller **180**.

Driving and operating completely independently of the drive motor **46** is a valve drive assembly **190**. The valve drive assembly **190** includes a variable speed electric drive motor **192** providing energy to a drive shaft **194** and a tone wheel **196** having a single or multiple set of teeth **198** from which the position or phase of the shaft **194** may be deduced. A sensor **202** such as a Hall effect variable reluctance sensor provides a signal through a line **204** to the controller **180**. Given the signals from the two phase sensor wheels **176** and **202**, the controller **180**, when provided with operating programs and criteria, is capable of providing independent electrical power through the conductors **206** to the motor **46** and through the conductors **208** to the motor **192** thereby adjusting the speed and phase of the variable speed electric drive motors **46** and **192** to adjust the relative phase relationship between the pistons **16**A and **16**B and the inlet valve assemblies **50**A and **50**B and the outlet valve assemblies **100**A and **100**B.

Affixed to the shaft **198** is a belt drive pinion **212** which drives a timing belt or chain **214** which engages drive pinions **216**A and **216**B which are coupled to the rotary valve bodies **106**A and **106**B and rotate them together.

The drive shaft **194** terminates in a first bevel gear **222** which mates with the second bevel gear **88** secured to the shaft **86**. The shaft **86** is terminated by the pair of cams **76** which cooperate with the cam followers **78** to reciprocably drive the valve plates **58**A and **58**B of the respective inlet valve assemblies **50**A and **50**B. It will thus be appreciated that the independent electric drives to, first of all, the piston and cylinder assemblies **14**A and **14**B and, second of all, the inlet valve assemblies **50**A and **50**B and the outlet valve assemblies **100**A and **100**B allow phase independent operation of these components to improve and adjust the operating characteristics of the positive displacement pump **10'**.

Such independent control may be further expanded such that the pistons **16**A and **16**B, the inlet valve assemblies **50**A and **50**B and the outlet valve assemblies **100**A and **100**B are all independently driven and thus their speed, and more importantly their phase, relative to the other components may be independently adjusted. This additional capability requires the addition of a third electric drive motor and appropriate sensor and modification to the controller **180** of the second alternate embodiment positive displacement pump **10'** described directly above to accept a third sensor and provide a third output to the third drive motor. So configured, the drive motor **46** drives the pistons **16**A and **16**B, the drive motor **192** drives the inlet valve assemblies **50**A and **50**B and the third drive motor drives the rotary outlet valve assemblies **100**A and **100**B. As noted, the additional tone wheel and sensor is utilized to provide speed and phase information to the controller as will be readily appreciated. So configured, independent speed and phase control of the pistons **16**A and **16**B, the inlet valve assemblies **50**A and **50**B and the rotary outlet valve assemblies **100**A and **100**B may be achieved.

As shown in FIG. **1**, a positive displacement pump **10** according to the present invention will include an pper piston and cylinder assembly **14**A and a lower piston and cylinder assembly **14**B whose pistons **16**A and **16**B operate 180° out of phase. The two passageways **104**A and **104**B deliver fluid to a common outlet **122**. This means that the outflow may be characterized as a d.c. level with a superimposed fluctuation that can be described as (for the first cycle)

$$q_{net}\Big]_o^T = q_u\Big]_o^{T/2} + q_e\Big]_{T/2}^T \tag{1}$$

where T is the reciprocal of the driving frequency (f_d) and

$$\omega = f_d/2\pi. \tag{2}$$

Turning then to the operation of the pump and FIGS. **9**, **10**, **11** and **12**, as suggested by equation (1), the upper passageway **104**A will deliver the fluid to be pumped for the time period $nT \leqq t \leqq (2n+1)T/2$ and the lower passageway **104**B will deliver fluid during the period $(2n+1)T/2 \leqq t \leqq (n+1)T$. This delivery is powered by the forward advance of the respective pistons **16**A and **16**B and it is controlled by the angular position (ϕ) of the rotary outlet valves **106**A and **106**B illustrated in FIG. **1**.

FIG. **9** shows the timing diagram for the rotary outlet valve **106**A for the upper piston and cylinder assembly **14**A and for the conditions of maximum flow rate. The timing diagram for the lower piston and cylinder assembly **14**B is identical to that of FIG. **9** except it is shifted by 180°. This diagram is presented for the rotary position of ϕ_1 of the upper rotary outlet valve **106**A as a function of the crank angle $\theta_1(t)$ of the crankshaft **34**A. It is assumed, for the present discussion, that $\theta_1(t)$ is the linear function:

$$\theta_1(t)=\omega t \tag{3}$$

and that ω=constant. This restriction is modified in a subsequent section in order to gain enhanced performance of the pump **10** in HVAC heat exchanger applications.

FIG. **11** presents the crank angle position versus time of one rotation of the crankshaft showing a non-linear relationship therebetween.

The second element of the controlled flow rate condition of the pump **10** is that of the sliding plates **58**A and **58**B with respect to the fixed bars **54**.

FIGS. **4** and **5** show the top and end views of the sliding inlet valve assemblies **50**A and **50**B. The symbol ξ_B refers to the lateral position of the sliding plate **58**A. The upper passageways are closed for $\xi_B=0$; they are fully open for $\xi_B=-W_A$.

FIG. **10** shows the inlet valve plates **58**A and **58**B plate positions: $[\xi_B(t)]$, for the maximum flow rate condition. Once again, the independent variable is considered to be $\theta_1(t)$ for the present discussion.

The magnitude of the volume flow rate will be linearly proportional to ω given the condition that the channel is fully filled from the surrounding plenum P on each stroke. (The filling stroke for the upper piston **16**A is $\pi \leqq \theta \leqq 2\pi$ as shown in FIGS. **9** and **10**.) Given the inertia of the elements involved, the change in flow rate that results from a change in the rotational speed (ω) of the crankshaft **34**A is considered to represent a "slow" change of the operating condition.

If the sliding inlet valve plates **58**A and **58**B (ξ_B, ϵ_C) and the outlet valves **106**A and **106**B (ϕ_1, ϕ_2) are used to control the flow, then the delivered mass can be adjusted on a time scale of one-half cycle. That is, for ω=constant, ϕ_1 can be held at $\phi_1=\pi/2$ (i.e., the rotary outlet valve **106**A is closed) and $\epsilon_B=-W_A$ (i.e., the inlet valve plate **58**A is open) for a period that is longer than that shown in FIGS. **9** and **10**. For example,

$$\phi_1=\pi/2 \text{ for } 0 \leqq \theta < 30 \text{ degrees,}$$

and

$$\phi_1 \rightarrow 0 \text{ at } \theta=30 \text{ degrees}$$

and

$$\phi_1=\pi/2 \text{ at } \theta=150 \text{ degrees}$$

while

$$\epsilon_B=-W_A \text{ for } 0 \leqq \theta < 30 \text{ degrees,}$$

and

$$-\epsilon_B \rightarrow 0 \text{ at } \theta=30 \text{ degrees}$$

and

$$\epsilon_B \rightarrow -W_A \text{ at } \theta=150 \text{ degrees}$$

Hence, the time period for which material is delivered from the upper passageway **104**A is reduced (in terms of the crank angle) from nominally 0→180 degrees to nominally 30→150 degrees. The ingestion portion of the cycle: 180→360 degrees, is unchanged from that of the maximum flow rate condition. However, in the θ→30 degrees and the θ: 150→180 degree segments, the ingested fluid and/or material will be expelled from the channel and returned to the supply plenum P.

If a multiphase, i.e., fluid and particulate, mixture is to be pumped, and if it would be useful to stir the material in the plenum P in order to enhance the uniformity of the discharge mixture, then the fractional operation noted above will provide an added benefit to that of the pumping action. Specifically, as noted above, the ingested material in the 0→30 degree and the 150→180 degree segments participates in a stirring action within the plenum P.

By adjusting the fractional valve openings and the rotational speed (ω), an optimal combination of stirring and net flow rate can be achieved.

For a given angular speed ($d\theta/dt=\omega$) of the crankshaft **34**A, the pressure: P_{face} (t) on the forward face of the piston **16**A will have a characteristic signature for an air-only operation and a given air density. (The non-dimensional representation: $\{[P_{face}-P_{atm}]/\rho V_p^2\}$ would be expected to be relatively independent from ρ and V_p.) The presence of a dispersed phase, i.e., the powder paint, will require the pressure at the face of the piston **16**A to be larger than the air-only case. This is shown in terms of the control volume momentum equation (for a control volume that is bounded by the interior faces of the sliding valve plate **58**A, the lower surface of the cylinder wall **20**A, the face of the piston **16**A and the centerplane of the control valve: "c").

$$\vec{F}_s = \frac{d}{dt}\int_{c,v} \rho\vec{V}\,d\forall + \int \rho\vec{V}\vec{V}\cdot\hat{n}\,dA \tag{4}$$

where the x (streamwise) component of $\vec{F}$ is given as

$$F_x = (p_{face} - p_c)A_{face} - \int_{A_{wall}} \tau_w\,dA \tag{5}$$

and τ_W is the wall shear stress acting on the fluid. The latter term in (5) is expected to be small with respect to the first term (RHS) and it is expected to be negligibly influenced by the presence of the particulate matter.

Since the density of the powder is nominally 10^3 times that of the continuous phase-air, a modest volume concentration (≈1 percent) will cause a readily measured increase in p_{face} for a given rotational speed of the crankshaft **34**A. Also, as noted above, it can be expected that $p_{face}=f(C_p)$ where C_p is the concentration of the particles.

Industrial Application—Powder Paint

The suggested operating protocol for the controllable positive displacement pumps **10** and **10'** in a powder paint application is to:

i) open the sliding plate **58**A, and

ii) open the rotary outlet valve assembly **100**A at the beginning of the stroke of the piston **16**A: θ≈0, and monitor $P_{face}(t)$ at the beginning of the delivery stroke. Given a proper calibration environment and recorded data, one can infer the volume concentration from the measured p_{face} over, e.g., the first 20 degrees of motion of the crankshaft **34**A. Given this information, control circuitry can then coordinate—for $\theta>20$ degrees—the combined: a) closing of the rotary outlet valve **100**A and b) opening the sliding plate **58**A which will cause the previously ingested "charge" to be returned to the inlet plenum P. (The stirring effect will also clearly benefit the mixture homogeneity in the plenum P as noted above.)

The above described operational protocol addresses the two principal needs of the powder paint delivery hardware: i) to identify, and ii) to control the mass of the particulate material that is to be delivered to the applicator. The use of the upper/lower piston and cylinder configuration of FIG. **1** will result in a relatively continuous discharge of powder.

Industrial Application—HVAC Systems

Another application for the controllable positive displacement pump **10** is that of air delivery to the heating/cooling coils of an HVAC system. Of concern for such a system is the upstream propagation of flow noise. In particular, the parallel shear layers that are formed from the fixed inlet valves **50**A and **50**B are likely sources of acoustic noise.

This concern suggests that the forward (pressurizing) stroke of the pistons **16**A and **16**B be executed relatively faster than the filling (return) stroke. Various mechanical linkages which can execute such drive patterns exist. FIG. **12** presents a representative $V_p(t)$ pattern than would meet the desired objectives. (V_p=velocity magnitude of the piston).

A concomitant advantage of this $V_p(t)$ pattern is that the velocity of air through the heating/cooling coils will be larger than would be the velocity in a symmetric-drive pattern. Specifically, the larger the velocity, the greater will be the momentary heat transfer and the greater will be the time averaged heat transfer for a given area of heat exchanger. This benefit is in addition to the intrinsic benefit of the positive displacement pump **10** according to the present invention for such heat transfer applications. Specifically, by creating twice the cycle average velocity over one-half of the heat exchanger for one-half of the cycle time, and repeating this behavior for the other one-half of the heat exchanger for one-half of the cycle time, a greater heat transfer will be obtained as enhanced heat transfer will derive from both the larger temperature differences and the larger convection heat transfer properties of the higher speed flow.

The foregoing disclosure is the best mode devised by the inventor for practicing this invention. It is apparent, however, that methods incorporating modifications and variations will be obvious to one skilled in the art of positive displacement pumps. Inasmuch as the foregoing disclosure presents the best mode contemplated by the inventor for carrying out the invention and is intended to enable any person skilled in the pertinent art to practice this invention, it should not be construed to be limited thereby but should be construed to include such aforementioned obvious variations and be limited only by the spirit and scope of the following claims.

I claim:

1. A controllable, positive displacement pump comprising, in combination,

a drive motor,

a pair of cylinders each having a sidewall and a head,

a pair of pistons received in a respective one of said pair of cylinders,

a crankshaft driven by said drive motor and having a respective pair of cranks for driving said pair of pistons,

a pair of reciprocating inlet valves defining a pair of adjacent gratings disposed adjacent a respective one of said pair of cylinder sidewalls, and

a pair of rotary outlet valves disposed adjacent a respective one of said pair of cylinder heads.

2. The controllable positive displacement pump of claim **1** wherein said pair of cranks are disposed in diametric opposition.

3. The controllable positive displacement pump of claim **1** wherein said drive motor is operably coupled to said reciprocating inlet valves and said rotary outlet valves and drives said crankshaft and said valves in synchronism.

4. The controllable positive displacement pump of claim **1** further including a second drive motor for driving said reciprocating inlet valves and said rotary outlet valves.

5. The controllable positive displacement pump of claim **4** further including tone wheels associated said crankshaft and said rotary outlet valves, sensors associated with said tone wheels and a controller for driving said drive motor and said second drive motor, said controller adapted to adjust the phase of said crankshaft and said rotary outlet valves.

6. The controllable positive displacement pump of claim **1** further including a second drive motor for reciprocating said inlet valves and a third drive motor for rotating said outlet valves.

7. The controllable positive displacement pump of claim **1** wherein said pair of adjacent gratings include elongate openings and wherein one of said gratings reciprocates transversely to said elongate openings.

8. The controllable positive displacement pump of claim **7** wherein at least one of said gratings define trapezoidal cross sections.

9. The controllable positive displacement pump of claim **1** further including a at least one drive belt pinion and at least one driven belt pinion associated with said drive motor and said rotary valves and at least one timing belt received on said belt pinions.

10. The controllable positive displacement pump of claim **1** further including a bevel gear drive having an input driven by said drive motor and an output driving at least one cam associated with said reciprocating inlet valves.

11. A controllable, positive displacement pump comprising, in combination,

a drive motor,

a pair of cylinders each having a sidewall and an end,

a piston received in each of said pair of cylinders,

a crankshaft driven by said drive motor and having a pair of cranks for driving a respective one of said pair of pistons,

an inlet valve having a reciprocating grating associated with each of said pair of cylinder sidewalls, and

a rotary outlet valve disposed adjacent each of said pair of cylinder heads.

12. The controllable positive displacement pump of claim **11** wherein said drive motor is operably coupled to said inlet valves and said rotary outlet valves and drives said crankshaft and said valves in synchronism.

13. The controllable positive displacement pump of claim **11** further including a second drive motor for driving said inlet valves and said rotary outlet valves.

14. The controllable positive displacement pump of claim **13** further including tone wheels associated said crankshaft and said rotary outlet valve, sensors associated with said tone wheels and a controller for driving said drive motor and said second drive motor, said controller adapted to adjust the phase of said crankshaft and said rotary outlet valves.

15. The controllable positive displacement pump of claim **11** further including a second drive motor for reciprocating said inlet valves and a third drive motor for rotating said outlet valves.

16. A controllable, positive displacement pump, comprising, in combination,

a housing defining a pair of cylinders having sidewalls and ends,

a piston received in each of said cylinders, a crankshaft defining a pair of diametrically opposed cranks, one of said cranks operably coupled to a respective one of said pistons,

a reciprocating inlet valve having a pair of gratings disposed adjacent said cylinder sidewall and said cylinder and,

a rotary outlet valve associated with each of said cylinders and disposed adjacent said cylinder end, and

a drive assembly for providing energy to said crankshaft and said reciprocating inlet valves and said rotary outlet valves.

17. The controllable, positive displacement pump of claim **16** wherein said housing further defines a pair of plenums.

18. The controllable, positive displacement pump of claim **16** wherein said drive assembly includes a first drive motor for driving said crankshaft, a second drive motor for driving said reciprocating inlet valves and said rotary outlet valves, tone wheels associated with said first drive motor and said second drive motor, sensors associated with said tone wheels and a controller for driving said first drive motor and said second drive motor, said controller adapted to adjust the phase of said first drive motor and said second drive motor.

19. The controllable, positive displacement pump of claim **16** wherein said pair of adjacent gratings include elongate openings and wherein one of said gratings reciprocates transversely to said elongate openings.

20. The controllable, positive displacement pump of claim **16** wherein said drive assembly includes a drive motor driving said crankshaft and said valves in synchronism.

* * * * *